Fermilab

Particle Physics Division

Mechanical Department Engineering Note

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Title: ILC Cryo-module Vacuum Test End Caps

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Key Words: **ILC, Cryo-module, Vacuum, Pressure Vessel, End Cap, Stainless Steel 304L, Stress, Allowable Stress, Yield Strength.**

Abstract/Summary: Two almost identical end caps are designed by Don Mitchell to be used in ILC Cryo-module vacuum test. When reversed, they can also be used to test the 3rd harmonic module. Finite element analysis is done to make sure that the maximum stress is with in the allowable of the material. ASME *Boiler and Pressure Vessel Code* is checked to assure that the design conforms with the code.

Applicable Codes: ASME *Boiler and Pressure Vessel Code* (2004 Edition).

General Description of ILC Cryo-module Vacuum Test End Caps

Two almost identical end caps are designed by Don Mitchell to be used in ILC Cryo-module vacuum test. When reversed, they can also be used to test the 3rd harmonic module. The cylindrical parts are 1/4 inch thick, of length 250 mm and 350 mm, and of diameter 965 mm and 1089 mm, respectively. The end plates and flanges are 1 inch thick.

For our analysis, we put on end plates on both ends so that the end cap is itself a mini vessel. Because the two end caps are almost identical, we analyze only one. Here we analyze the upstream end cap. A simplified model is shown in Fig.1 (where a quarter is cut out to reveal the inside).

All material used for end cap is stainless steel 304L. We use following typical material properties in our analysis: Young's modules $E = 28e6$ psi, Poisson's ratio $n = 0.3$, Yield strength $s_y = 30e3$ psi, Allowable stress $s_a = 20e3$ psi.

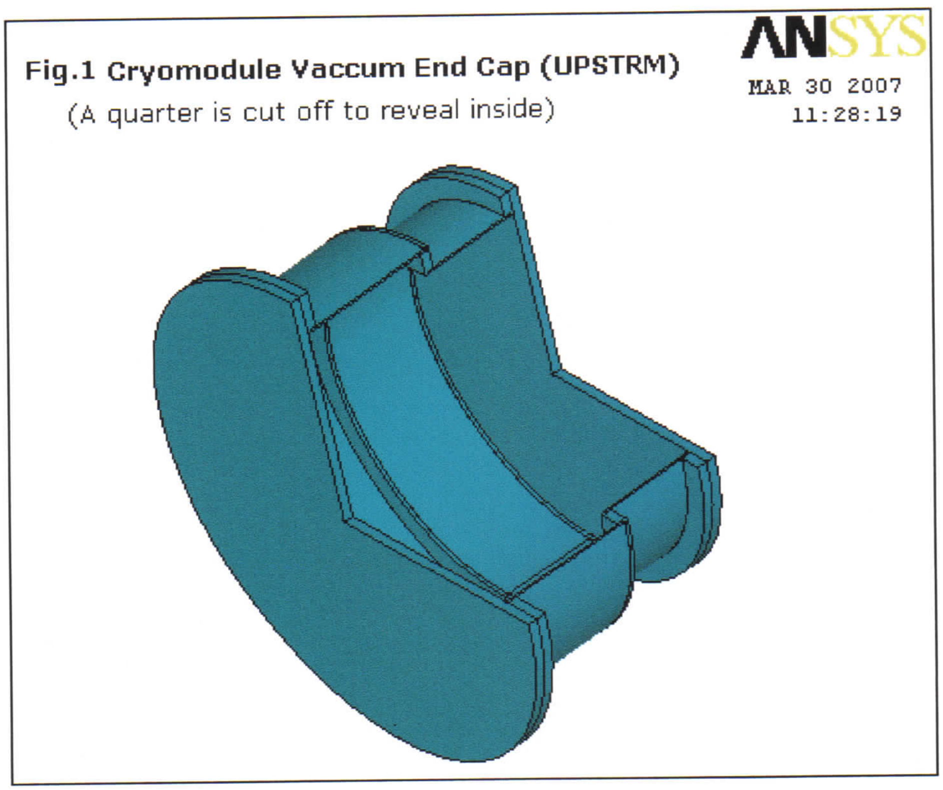
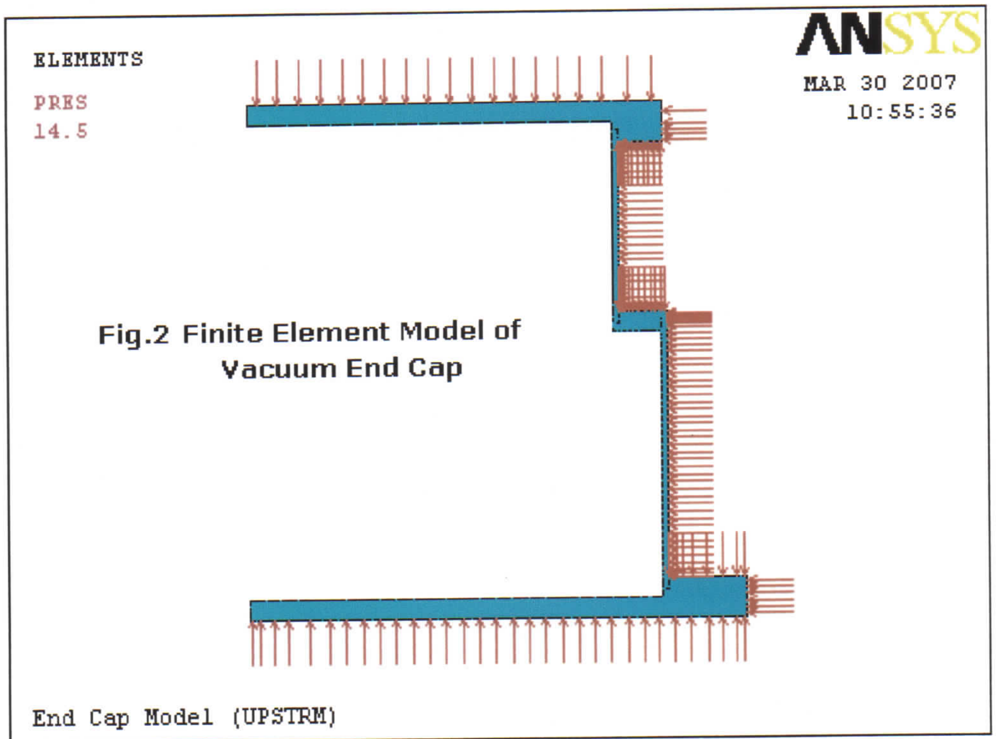
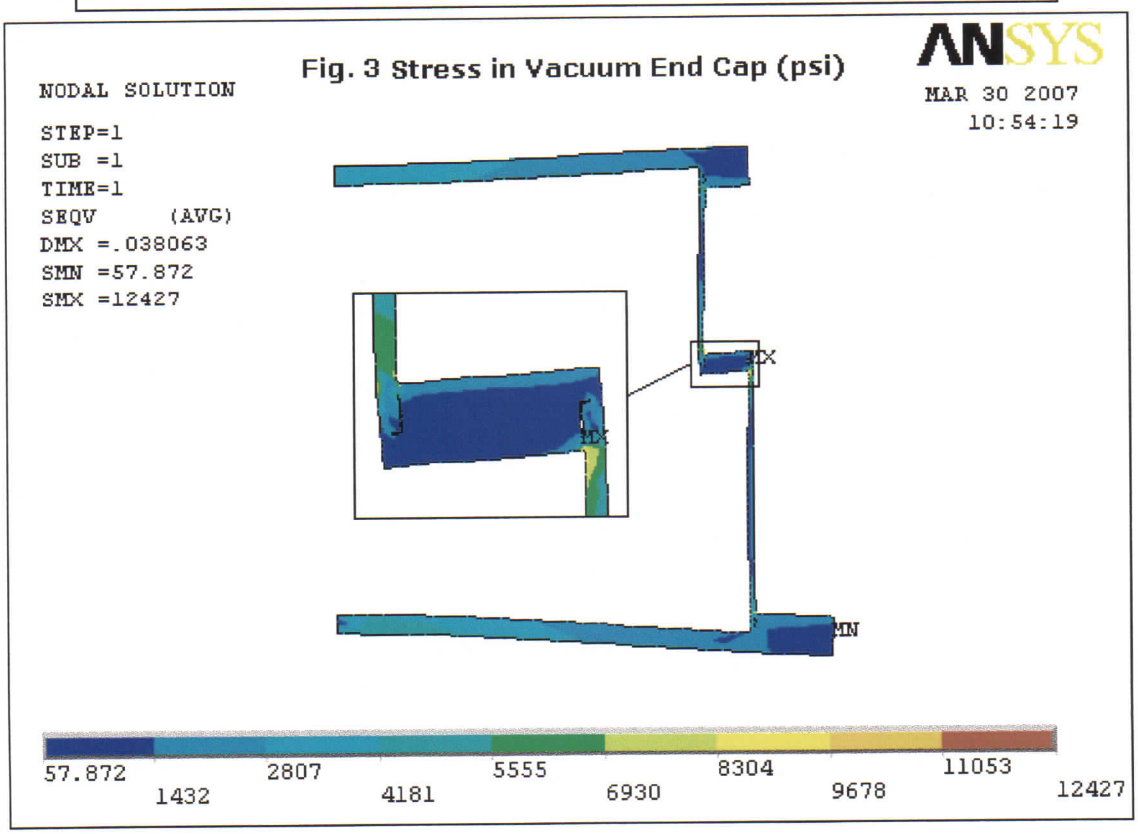


Fig.1 Cryomodule Vaccum End Cap (UPSTRM)
(A quarter is cut off to reveal inside)

Finite Element Analysis of ILC Cryo-module Vacuum Test End Caps

A simple 2-d axial symmetric model is shown in Fig.2. The result is plotted in Fig.3.

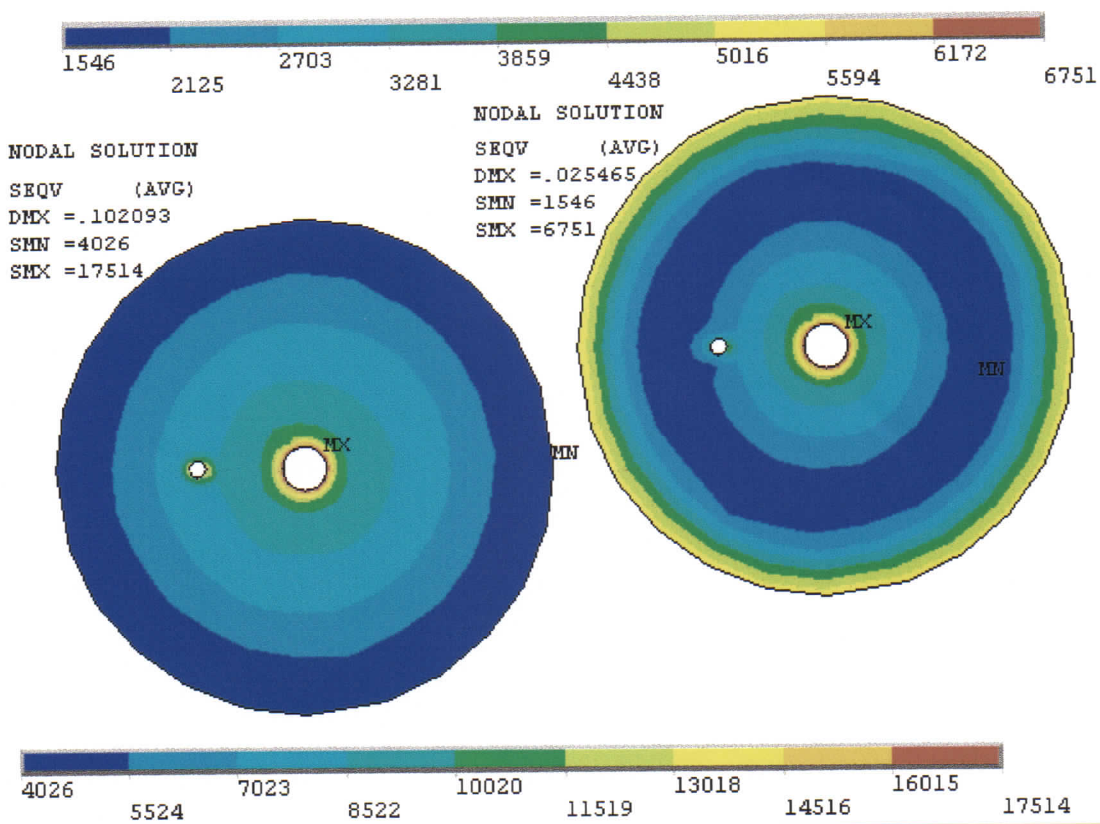




FEA model shows that the maximum stress is 12 ksi, while the allowable stress of the material is 20 ksi. The maximum displacement is about 1 mm.

In above FEA model, the holes in end plate are omitted. To see the effect of the holes, we made a separate model of end plate, as shown in Fig.4. The pressure forces are added at the edge of the holes. As for outside edge, in one case we applied simple support, in another we used moment support (no rotation). We have maximum stress of 17514 psi for simply supported plate, and 6751 psi for moment supported plate. The real situation is between the two cases provided in the model, as can be seen from Fig.3. Furthermore, the half nipples were omitted in the model. When they are added, the stress level will be lower. Therefore, the stress in the end plate with holes is still reasonably small.

Fig.4 Stress in End Plate with Holes



Check the design with ASME *Boiler and Pressure Vessel Code*

1. Tubes

Volume VIII-2, Section AD 310.1 Shell of revolution under external pressure (pg 81).

1)	Thickness	$t = 6.35$ mm (1.3 GHZ),	6.35 mm (3.9 GHZ)
	Length	$L = 250$ mm,	350 mm,
	Diameter	$D = 965$ mm,	1089 mm,
		$L/D = 0.259$,	0.321,
		$D/t = 152$,	171.

2) From Volume II-D, Fig G in subpart 3 (pg706), we use $L/D = 0.25$, $D/t = 150$,
3) we get factor $A = 0.03$.
4) From Fig. HA-3 (same volume, pg 713), for stainless steel 304L under room temperature,
5) we get factor $B = 11000$.
6) Allowable load

$$P_a = \frac{4B}{3(D/t)} = \frac{4\times 11000}{3\times 150} = 97.8 \text{ psi.}$$

The actual load is 14.5 psi, we have a safety factor of 97.8/14.5 = 6.7. Therefore cylindrical shells are okay.

2. Cover Plates

Volume VIII-2, Section AD 702 (b), The minimum thickness of cover plate (pg110).

$$t = d\sqrt{CP/S + 1.9Wh_g/Sd^3}$$

Diameter d = 1000 mm = 39.4 inch. From Fig. AD-701.2, parameter $C = 0.3$. Load P = 14.5 psi. From Table 1A of volume II-D (pg 84), Allowable stress for stainless steel 304L plate S = 20000 psi. W is the total bolt load, we assume it to be the product of pressure and the cover plate size $W = P\,(\pi d^2/4)$ = 14.5(1219)=17679 lb. Let parameter h_G = 2 in. We have

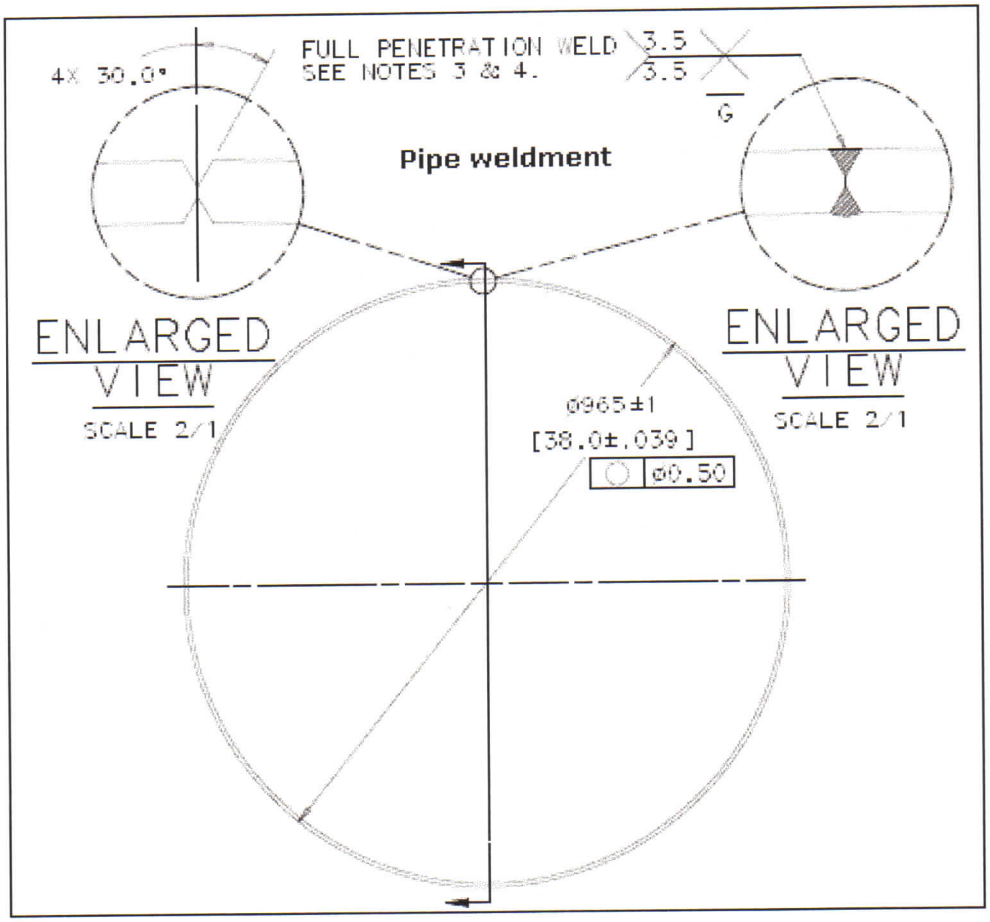
$$t = 39.4\sqrt{0.3\times 14.5/20000 + 1.9\times 17679\times 2/20000\times 39.4^3} = 0.65 \text{ in.}$$

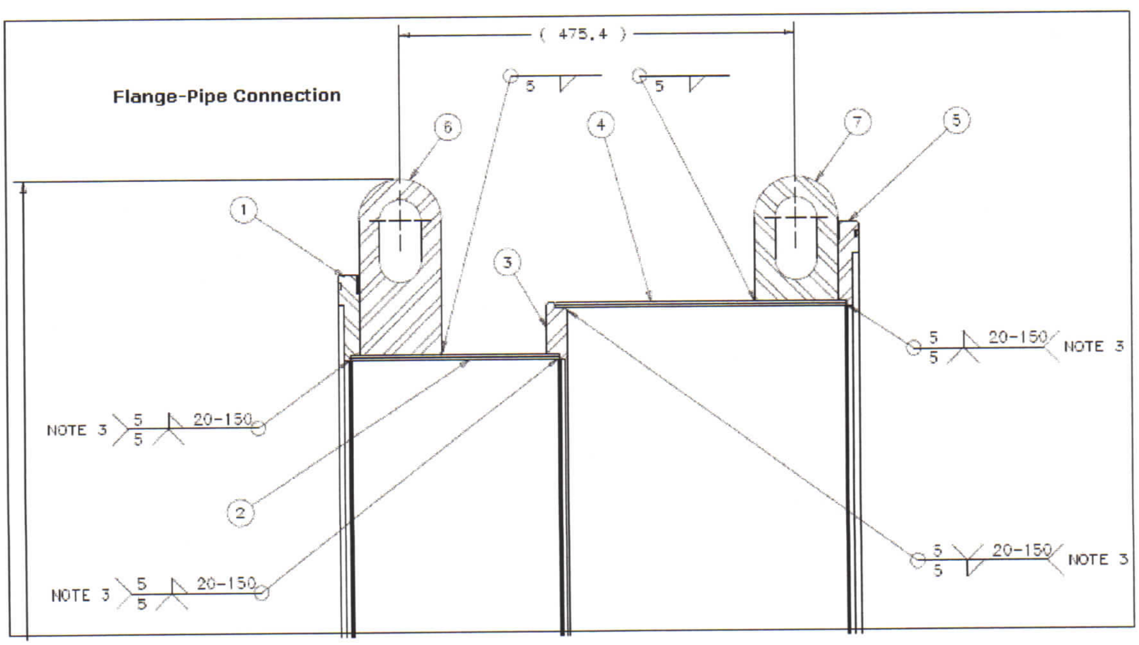
Actual thickness is 1 in. So cover plates are okay.

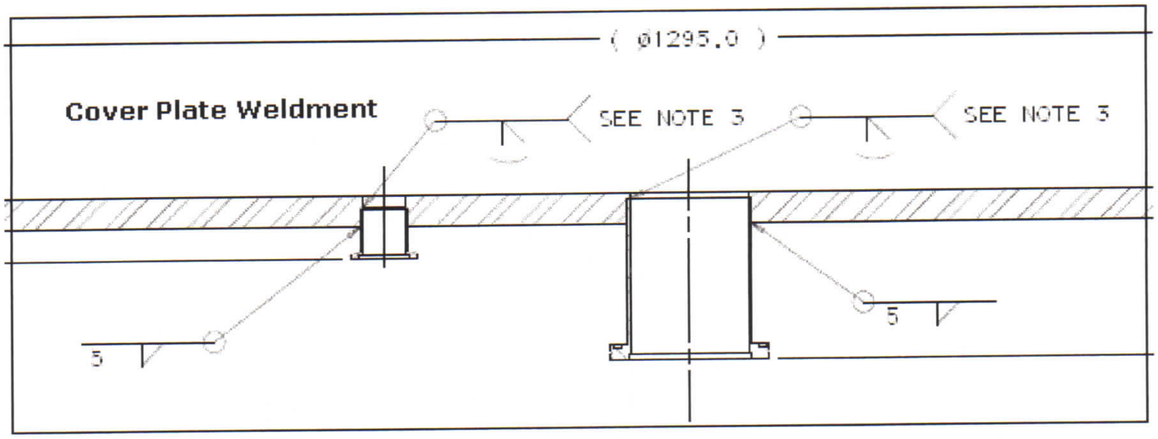
3. welds

Some typical welds are shown in Appendix, they are checked with Article D-4 of Volume VIII-2. No problems were found, so they are okay.

Appendix Typical Weldments



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Cover Plate Weldment
SEE NOTE 3
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